10/625014 Copy

I hereby certify that this correspondence is being deposited with the United States Postal Service as first class mail in an envelope addressed to: Commissioner for Patents, P.O. Box 1450 Alexandria, VA 22313 on May 30, 2006.

James S. Parker, Patent Attorney

REQUEST FOR CERTIFICATE OF CORRECTION UNDER 37 CFR 1.322
Docket No. RTI-101XC1
Patent No. 7,010,936

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

Applicants	:	Daniel P. Rini, Louis Chow, H. Randolph Anderson, Jayanta Sankar Kapat, Bradley Carman, Brian Gulliver, Jose Mauricio Recio
Issued	:	March 14, 2006
Patent No.	:	7,010,936
For	:	Method and Apparatus for Highly Efficient Compact Vapor Compression Cooling

Mail Stop Certificate of Corrections Branch
Commissioner for Patents
P.O. Box 1450
Alexandria, VA 22313-1450

REQUEST FOR CERTIFICATE OF CORRECTION
UNDER 37 CFR 1.322 (OFFICE MISTAKE) AND CFR 1.323 (APPLICANT'S MISTAKE)

Sir:

A Certificate of Correction (in duplicate) for the above-identified patent has been prepared and is attached hereto.

In the left-hand column below is the column and line number where errors occurred in the patent. In the right-hand column is the page and line number in the application where the correct information appears.

Patent Reads:	**Application Should Read:**
Column 2, lines 45-47:	Page 3, lines 19-20:
"300 Watts of heating while consuming less than about 100 Watts of electrical power"	--300 watts of heating while consuming less than about 100 watts of electrical power--
Column 4, line 42:	Page 6, line 23:
"with the subject invention"	--with the subject invention.--
Column 4, line 62:	Page 7, line 8:
"epiterchoid"	--epitrochoid--
Column 6, line 9:	Page 9, line 3:
"users body while allowing"	-- user's body while allowing --
Column 6, lines 11-12:	Page 9, lines 4-5:
"against a users body"	--against a user's body--
Column 6, line 28:	Page 9, line 16:
"show in FIGS. 11A and 11B"	--shown in FIGS. 11A and 11B--
Column 6, line 45-46:	Page 9, line 28-29:
"about 300 Watts of heat while consuming less than about 100 Watts"	--about 300 watts of heat while consuming less than about 100 watts--
Column 8, line 29:	Page 12, line 20:
"epiterchoid"	--epitrochoid--
Column 8, line 42:	Page 12, line 26:
"an epiterchoid shape"	--an epitrochoid shape--
Column 8, lines 53-54:	Page 13, line 5:
"epiterchoid"	--epitrochoid--

Column 8, lines 55-56: "epiterchoid" — Page 13, line 7: --epitrochoid--

Column 8, line 61: "epiterchoidal" — Page 13, line 10: --epitrochoidal--

Column 9, line 22: "In embodiment the second " — Page 13, line 29: --In an embodiment, the second--

Patent Reads:	**Preliminary Amendment dated 7/20/04 Reads:**
Column 9, lines 45-46:	Page 3, line 6:
"of to compressor housing"	--of the compressor housing--

Patent Reads:	**Application Should Read:**
Column 10, line 17:	Page 15, line 12:
"in order maximize"	--in order to maximize--
Column 10, line 33:	Page 15, line 21:
"epiterchoidal"	--epitrochoidal--
Column 10, lines 42-43:	Page 15, lines 27-28:
"low friction, low corrosion materials"	--low friction and low corrosion materials--
Column 12, line 62:	Page 19, lines 17-18:
"To remove 200 Watts of heat"	--To remove 200 watts of heat--
Column 13, line 49:	Page 20, line 25:
"given number of times, N, given by"	--given number of times, *N*, given by"

JUN 8 2006

Column 13, line 57:

"where n is the number"

Page 20, line 26:

--where *n* is the number--

Fifth column of table at column 14, line 6:

"N"

Fifth column of table at page 21, line 7:

-- *N*--

Column 16, lines 44-45:

"An archemidian spiral"

Page 25, line 4:

--An Archemidian spiral--

Column 16, line 51:

"where the constants A and B"

Page 25, line 7:

-- where the constants *A* and *B*--

Patent Reads:

Amendment dated 8/26/05 Reads:

Column 18, line 65:

"An apparatus for cooling comprising"

Page 4, Claim 9, Line 1:

-- An apparatus for cooling, comprising --

Column 19, line 21:

"surface of the condensor"

Page 5, Claim 9, lines 8-9:

--surface of the condenser--

Column 19, line 24:

"the heat surface"

Page 5, Claim 9, line 11:

--heat transfer surface--

Patent Reads:

Application Should Read:

Column 20, line 19:

"is throttling valve"

Page 31, Claim 21, line 2:

--is a throttling valve.--

Column 21, line 8:

"epiterchoid"

Page 32, Claim 27, line 4:

--epitrochoid--

Patent Reads:

Amendment dated 8/26/05 Reads:

Column 21, line 34:

"for cooling comprising"

Page 10, Claim 36, line 1:

--for cooling, comprising--

Patent Reads:

Column 24, line 16:

"is a helical duet."

Application Reads:

Page 37, Claim 59, line 2:

--is a helical duct.--

Patent Reads:

Column 24, line 28:

"enhanecd surface geometry"

Amendment dated 12/13/04 Reads:

Page 22, Claim 97, line 3:

"enhanced surface geometry"

A true and correct copy of pages 3, 6, 7, 9, 12, 13, 14, 15, 19, 20, 21, 25, 31-32, 37, of the specification, a copy of page 3 of the Preliminary Amendment dated July 7, 2004, a copy of page 22 of the Amendment dated December 13, 2004, and a copy of pages 4, 5, and 10 of the Amendment dated August 28, 2005, as filed, which support Applicants' assertion of the errors listed, accompany this Certificate of Correction.

The Commissioner is authorized to charge the fee of $100.00 for the amendment to Deposit Account No. 19-0065. The Commissioner is also authorized to charge any additional fees as required under 37 CFR 1.20(a) to Deposit Account No. 19-0065. Two copies of this letter are enclosed for Deposit Account authorization.

Approval of the Certificate of Correction is respectfully requested.

Respectfully submitted,

James S. Parker
Patent Attorney
Registration No. 40, 119
Phone No.: 352-375-8100
Fax No.: 352-372-5800
Address: P.O. Box 142950
Gainesville, FL 32614-2950

JSP/lkw

Attachments: Copy of pages 3, 6, 7, 9, 12, 13, 14, 15, 19, 20, 21, 25, 31-32, 37, of the specification, a copy of page 3 of the Preliminary Amendment dated July 7, 2004, a copy of page 22 of the Amendment dated December 13, 2004, and a copy of pages 4, 5, and 10 of the Amendment dated August 28, 2005

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microclimate systems. It is extremely difficult to find a commercially available compressor alone which is smaller than 1 liter and weighs less than several pounds, and which is rated for a cooling load near or below 500 watts. The cycle would then need additional components such a condenser and evaporator to become effective.

Accordingly, there is need for a cooling system having a high coefficient of performance and a light compact design.

Brief Description of the Invention

The subject invention pertains to a method and apparatus for cooling. In a specific embodiment, the subject invention relates to a lightweight, compact, reliable, and efficient cooling system. The subject system can provide heat stress relief to individuals operating under, for example, hazardous conditions, or in elevated temperatures, while wearing protective clothing. The subject system can be utilized in other applications that can benefit from this type of cooling system. The performance of this system cannot be matched simply by using smaller versions of currently available designs. In a specific embodiment, the subject microclimate system can remove at least about 120 watts of heat while consuming less than about 50 watts of power, and weigh less than about 2.5 pounds while having less than about a 1000 cubic centimeter volume. In a further specific embodiment, the subject cooling system can remove at least about 300 Watts of heat while consuming less than about 100 Watts of electrical power, and can weigh less than about 3.5 pounds (not including the water jacket or the power source) within a volume of less than about 1500 cc or 1.5 L. In a specific embodiment, the subject system can run for at least about 4 hours or more with the use of batteries.

In a specific embodiment, the subject invention pertains to a cooling system having a total weight of less than about 3.5 pounds, a coefficient of performance of at least 2.4, and a volume of less than about 1500 cc with a cooling capacity between about 100 and about 500 watts. The subject cooling system can provide between 28 and 140 watts of cooling per pound and occupy between 3 and 15 cc of volume per watt of cooling. In comparison, commercially available units for cooling in this range would provide between 2.7 and 18.5 watts of cooling per pound and occupy a volume of

Brief Description of the Drawings

Figure 1A shows an embodiment of the subject invention.

Figure 1B shows an expanded view of a compressor incorporated with the embodiment shown in Figure 1A

Figure 2 shows a view of the interior of an embodiment of the subject invention, illustrating an annular region for hot vapor coolant flow and pin fins in thermal contact with the outer wall of the annular region.

Figure 3 shows an embodiment of an evaporator in accordance with the subject invention.

Figure 4 shows an embodiment of the subject invention showing a view of the interior of an embodiment of the subject invention, illustrating a pump, a motor, and a motor controller.

Figure 5 shows an embodiment of the subject invention, illustrating connections between various parts which allow liquids and/or gases to enter and/or exit the various parts.

Figure 6 shows an exploded view of a specific embodiment of a compressor in accordance with the subject invention.

Figures 7A and **7B** show two views of a specific embodiment of an evaporator in accordance with the subject invention.

Figure 8A shows an inner wall piece with a spiral spacer and an outer wall piece with pin fins of a specific embodiment of a condenser in accordance with the subject invention

Figure 8B shows the condenser shown in Figure 8A with the inner wall piece inserted into the outer wall piece to form a refrigerant annulus.

Figure 9A shows a schematic of a cooling system in accordance with the subject invention, incorporating a condenser, an expansion valve, an evaporator, and a compressor.

Figure 9B shows a basic vapor compression cycle temperature/entropy diagram.

Figure 10 shows the cross-section of a fin design for a compressor in accordance with the subject invention.

Figure 11A shows an embodiment of the subject invention having two fans and the battery within the condenser inner walls.

Figure 11B shows a cross section of the embodiment shown in Figure 11A, showing a "peanut" shaped cross section of the condenser walls with the battery, compressor motor, and evaporator within the inner condenser walls.

Figure 12 shows an example of epiterchoid shape, which a compressor chamber can incorporate in a specific embodiment of the subject invention.

Figure 13 shows an Archemidian spiral corresponding to a fluid path within an evaporator in accordance with a specific embodiment of the subject invention.

Detailed Description of the Invention

The subject invention pertains to a method and apparatus for cooling. In a specific embodiment, the subject invention relates to a lightweight, compact, reliable, and efficient cooling system. The subject system can provide heat stress relief to individuals operating under, for example, hazardous conditions, or in elevated temperatures while wearing protective clothing. The subject system can be utilized in other applications that can benefit from this type of cooling system. The performance of this system cannot be matched simply by using smaller versions of currently available designs.

The subject invention also relates to a condenser for transferring heat from a refrigerant to an external fluid in thermal contact with the condenser. The subject condenser can have a heat transfer surface and can be designed for an external fluid, such as air, to flow across the heat transfer surface and allow the transfer of heat from heat transfer surface to the external fluid. In a specific embodiment, the flow of the external fluid is parallel to the heat transfer surface. In another specific embodiment, the heat transfer surface can incorporate surface enhancements which enhance the transfer of heat from the heat transfer surface to the external fluid. In another specific embodiment, an outer layer can be positioned above the heat transfer surface to create a volume between the heat transfer surface and the outer layer through which the external fluid can flow.

Such a flow path can allow a user to conveniently wear the subject cooling system on the user's body as the flowing air exits the subject cooling system to be directed parallel to the users body while allowing intake of air at the first end unobstructed by the user. In a specific embodiment, the tubular condenser can be contoured to lie against a users body and can house the remaining components of the cooling system within a volume created by an inner surface **800** of the condenser. Figures 11A and 11B show an embodiment of the subject cooling system where the battery, compressor, motor, water pump, and evaporator are housed within the condenser, in a volume created by the inner surface **800** of the condenser. In this embodiment, Figure 11A shows a cross section from the top and Figure 11B shows a cross section from the side. As shown in Figures 11A and 11B, the fans produce a flow of air which travels through the shell, or annular volume, of the condenser formed between the heat transfer surface **880** of the condenser and an outer wall, or outer layer **10**, of the condenser. Another portion of the flowing air produced by the fans can travel through the portion of the condenser housing the battery, compressor, motor, and evaporator and remove heat from these components. In the embodiment show in Figures 11A and 11B, the compressor, motor, evaporator, and battery are each cylindrical in shape. Other shapes for one or more of these components can also be used.

The use of cylindrical components as shown in Figures 11A and 11B can also enable the use of a condenser with a substantially cylindrical shape with the battery within the same cylindrical volume as the compressor, motor, and evaporator. Alternatively, one or more components, such as the battery can be outside of this volume created by the condenser. In addition, a portion of one or more components can extend out from the volume created by the condenser.

In a specific embodiment, the subject microclimate system can remove at least about 120 watts of heat while consuming less than about 50 watts of power, and weigh less than about 6 pounds while having less than about a 1000 cubic centimeter volume. In a further specific embodiment, the subject cooling system can remove at least about 300 Watts of heat while consuming less than about 100 Watts of electrical power, and can weigh less than about 3.5 pounds (not including the water jacket or the power source)

can start with a certain volume of refrigerant vapor and reduce the volume by a set amount resulting in compressed refrigerant vapor. The amount of volume change can be a function of the geometry of the positive displacement means. Valves and upstream conditions typically govern the pressure at which the vapor leaves the compressor. The positive displacement means can be, for example, a piston style, a sliding vane, a screw, a scroll, or a rotary lobed type. In a specific embodiment, compressor **515** can incorporate a rotary lobed type positive displacement means. An example of this type of compressor is shown in Figures 1 and 6, and can be referred to as a rotary lobed compressor. The purpose of the compressor is to intake low pressure, low temperature refrigerant vapor and discharge high temperature high pressure vapor to the condenser.

Referring to Figures 1 and 6, the configuration shown can be referred to as a Wankel compressor. The compressor can incorporate a substantially triangular shaped rotor **624** which spins on an eccentric shaft **634**. In a specific embodiment, the compressor can use a 3/2 gear ratio for positioning (Ogura, Ichiro, "The Ogura-Wankel Compressor—Application of a Wankel Rotary Concept as Automotive Air Conditioning Compressor," SAE Technical Paper 820159, SAE 1982). The gears **632** are used to position the rotation of the rotor through its eccentric path. The rotor rotates inside of a peanut shaped epitrochoid chamber **626**. Such a rotor positioning results in the compressor exhibiting two complete compressions per revolution.

The shape of an epiterchoid chamber is determined by the following equations:

$$x(t) = \frac{3}{7}\cdot MA\cdot\cos(t) - \frac{1}{14}\cdot MA\cdot\cos(3MA\cdot t)$$

$$y(t) = \frac{3}{7}\cdot MA\cdot\sin(t) - \frac{1}{14}\cdot MA\cdot\sin(3MA\cdot t)$$

where MA is the major axis.

In a specific embodiment, a length of 49 mm can be utilized for the major axis of the epitrochoid with a height of 6 mm. Using the above equations, an epiterchoid shape, which is framed in a Cartesian coordinate system, is found to have the shape shown in Figure 12. The values of the major axis and height can be modified based on the cooling capacity requirements of the vapor compression cycle and the desired angular velocity of

the compressor. Once these two constraints are set, the basic designs of the main components of the compressor can be determined as a function of the geometry. The major axis determines the size of the rotor and the shape of the epitrochoid, as well as the gears that are used in the compressor.

Using the equations relating to the shape of the epiterchoid chamber suggested above, the rotor size and shape can also be chosen. Finally, the geometric height of the epiterchoid and rotor can be determined by the amount of fluid that is desired to be displaced on each revolution. After having calculated these dimensions, the compressor's speed can be chosen to determine the displacement per unit time or volumetric flow rate. In a specific embodiment, incorporating an epiterchoidal chamber with a major axis of 49 mm and a height of 6 mm, a speed of 1200 rpm is chosen to provide a mass flow rate of approximately 1 g/s of vapor refrigerant 134a at an inlet pressure of 57 psia.

The flow through the compressor can be controlled by inlet port **517** (shown in Figures 5 and 6) and valved exhaust ports **629** (shown in Figure 6). In a specific embodiment, a triangular inlet port **517** design based on the rotational path of the rotor can be used on the bottom face of the compressor. Although a triangular shaped port is shown here, other shapes such as oval, round, and square can also be used. This design can allow the cool refrigerant vapor into the compressor. Rotor **624** can then travel over the top of the intake port so as to close the intake port as rotor **624** begins to compress the refrigerant vapor. This design feature can eliminate the need for an intake check valve, typically used by positive displacement compressors. Exhaust valve **618** and valve stop **616** can be placed on the top face of the compressor and positioned on top of the exhaust port **629** to allow for the maximum compression to occur. The exhaust valve is a check valve that can prevent hot high pressure refrigerant vapor from flowing backwards into the compressor. In a specific embodiment, cantilevered flapper valves can be used to reduce the amount of space required for the outlet port **629**.

To reduce the vibrations caused by the mass of the rotor spinning eccentrically in the compressor, a counter balance **635** can be placed on the main shaft. A second rotor can be used to balance the compressor. In embodiment the second rotor can be positioned 180 ° out of phase with the first rotor so as to counter balance the rotating force. The

between the chambers during the rotary motion of the rotor. In a specific embodiment, the seals can be made of a low friction material to minimize wear and friction losses. In a further specific embodiment, an engineered plastic material such as PEEK, TEFLON, NYLON, or DELRIN can be used. Other materials with similar characteristics can also be used. The tip seals and face seals are spring loaded to insure that the plastic seals stay in contact with the metal surfaces of the compressor housing. In a specific embodiment, the springs used are 2.4 mm in diameter, 6.2 mm long, have a spring stiffness constant of 2.2 lbs per inch, and a pitch of 35 coils per inch. Preferably, at least one spring is used on each of the tip seals. Multiple springs can be used on the face seal in order to provide an even spring loading force. In further embodiments, the spring force can be produced by other means such as wave springs, elastic rubbers, or gas filled balls. Preferably, the tip and face seals are fabricated so that a slip fit into the rotor can be maintained. In a specific embodiment, a slip fit dimensional tolerance of 8 micron is used.

Please replace the paragraph at page 16, lines 7-26 with the following amended paragraph:

The motor **513**, as shown in Figure 1A, can be used to power the drive shaft **514.** In a specific embodiment, motor **513** can be a permanent magnetic synchronous motor. Other mechanical devices capable of producing shaft power can also be used to power the subject compressor, including, for example, combustion engines, wind, or paddlewheels. In a specific embodiment, the motor can be designed for long service life and can operate at much higher efficiencies than standard motors. The motor design can be a compact unit specially suited for this type of application. The motor can deliver a high power density and operate at variable speeds through a motor controller **23**. The incorporation of motor controller **23** can allow the motor to change the amount of compression, depending on the cooling load. Standard vapor compression cycles typically turn the compressor on and off in order to adjust to the net cooling requirements of the cooling load. The turning of the compressor on and off can reduce the efficiency of the cooling system, as the start up interval of a motor can be extremely inefficient. Accordingly, the use of a control feature,

compressor. Cooling fins **636** can be designed to increase the surface area of the outside housing to improve heat transfer out of the compressor housing. Cooling fins **636** can have a variety of shapes. In a specific embodiment, the cooling fins **636** can have long narrow channels running axially with the compressor. During operation of the subject cooling system, air can be blown past the compressor housing to help cool the internal components. In a specific embodiment, air flow provided by the condenser fan **570** can flow between the condenser inner wall surface **800** and the compressor **515** outer wall in space **900**, for example as shown in Figure 5. This air then comes in contact with the compressor cooling fins **636.** The number of fins and the size and shape of the fins can be chosen to enhance the cooling effect provided by air flowing over the fins. In one example, the number and size of the fins are chosen to be 48 and 0.25 inches, respectively, in order maximize the Nusselt number of the fluid flowing past the fins. The Nusselt number is directly proportional to the amount of heat transfer between the solid surface and the fluid and is known as:

$$Nu = 1.86 \cdot \left(\frac{Re \cdot Pr}{\frac{w}{D_h}} \right)^{\frac{1}{3}} \cdot \left(\frac{\mu}{\mu_s} \right)^{.14}$$

Where Re is the Reynolds number, Pr is the Prandtl number, w is the channel width, D_h is the hydraulic or effective diameter, μ is the bulk fluid viscosity, and μ_s is the fluid viscosity at the heat transfer surface.

For a specific embodiment of a compressor in accordance with the subject invention incorporating an epiterchoidal chamber with a major axis of 49 mm, a cross-sectional geometry shown in Figure 10 was chosen.

This direct cooling of the compressor can aid in the thermodynamic cycle shown in Figure 1, by reducing the superheat of the vapor between points 2 and 2s. Typical vapor compression cycles remove the heat from the compressor via the internal flow of the refrigerant. This increases the heat load of the vapor compression cycle and reduces cycle efficiency. The subject compressor can incorporate low friction, low corrosion materials. In addition, wear parts other than the seals can be coated with low friction,

between about 200 and about 500 square centimeters, with a surface area increase due to extended surfaces of 2 to 5 times, can provide up to 300 watts of cooling. In a further specific embodiment, the bases area is between about 300 and about 400 square centimeters with a surface area increase due to extended surfaces of 2.5 to 4 times and providing between 200 and 250 watts of cooling.

In a specific embodiment, extended surface features **860** can have an elliptical cross section. The elliptical cross section can provide a reduced pressure loss (allowing more air flow) so as to increase h. Examples of the utilization of extended surfaces having elliptical cross sections is given in Li, Q., Chen, Z., Flechtner, U., and Warnecke, H.J., "Heat Transfer and Pressure Drop Characteristics in Rectangular Channels with Elliptic Pin Fins," *Heat and Fluid Flow* 19 (1998) 245-250, which is hereby incorporated by reference. These extended surfaces can then be placed on the outside of the cylindrical cooling device in, for example, a staggered arrangement. Referring to Figure 8B, in a specific embodiment the extended surfaces can be placed with spacing **884** (in a direction parallel with the flow of air) and spacing **882** (in a direction perpendicular to the flow of air) set to, for example, 2.5 times the equivalent diameter of the ellipse. In a specific embodiment, the length of the elliptical pin is 1.66 cm. To remove 200 Watts of heat, fins **860** with an equivalent diameter of 4.19 mm can be used. An airflow device **570** can be placed at one end of the cylinder to flow air axially past the extended surfaces.

Accordingly, heat can be transferred between the hot compressed vapor refrigerant and an external fluid. In a specific embodiment, heat is transferred from the hot compressed vapor refrigerant to an ambient fluid, such as air or water, on the refrigerant side of the heat exchanger. This heat transfer can involve, for example, a simple flat plate, straight tubing, or a coil of tube that flows the condensing fluid by an air-cooled or liquid-cooled surface. In specific embodiments, condensing fluid can flow through a simple annulus or cylindrical design with a open path from top to bottom, through a series of straight ducts created within the annulus or cylinder, or through one or more spiral wound ducts created around the inside of the annulus or cylinder. The heat removal from the coil can also be calculated by $q = hA\Delta T$ where q [W] is the heat removal, h[W/m^{2}K] is the heat transfer coefficient, A[m^2] is the surface area of the cooled

surface, and ΔT [K] is the temperature difference between the cooled surface and the refrigerant. The temperature of the refrigerant can drop until it begins to condense, at which point it can remain at a constant temperature until the refrigerant is fully condensed into liquid.

In a specific embodiment, a condenser in accordance with the subject invention can incorporate one or more helical ducts created, for example, by a spiral wound wire tube **890** (shown in Figures 2 and 4) or an annulus **840** cut into an insert **810** (shown in Figures 5 and 8A). There can be one, or a plurality, n, channel(s) which transport the refrigerant from one end of the condenser to the other end of the condenser. In a specific embodiment with a plurality of channels, each channel can begin at a first end of the condenser and travel parallel to the other channels to the other end of the condenser. In a further specific embodiment, the plurality of parallel channels can spiral from one end of the condenser to the other end such that the refrigerant can travel slower in each channel to traverse the condenser. Referring to Figures 8A and 8B, insert, or first element, **810** is inserted into an outer piece, or second element, having dividing wall **870** from which surface extensions **860** extend from heat transfer surface **880**, such that lips **850** contact dividing wall **870** to seal the windings of annulus **840** from each other. Vapor refrigerated within the ducts can be in thermal contact with the dividing wall **870**. A cylindrical shape can enhance the amount of surface area available for a given volume. The duct can wrap around in a spiraling shape from the top of the cylinder to the bottom. In a specific embodiment, the shape of the tube, annulus, can be rectangular, in order to increase the surface area of the tube walls in contact with the hot vapor refrigerant. In this embodiment, the perimeter of the annulus is $P=2(w+y)$ where w is the width of the channel, or duct, and y is the height. Each channel wraps around the cylinder a given number of times, N, given by $N = \frac{L_{channel}}{\pi d}$, where d is the diameter of the cylinder. Since $L_{channel} = f(P,n) = f(w,y,n)$, therefore, $N = f(w,y,n)$, where n is the number of parallel channels wrapping around the cylinder such that refrigerant flows through each of the parallel channels, simultaneously, from the first end of the condenser to the second end of

the condenser. Therefore, the length of the coil, assuming 1mm thickness between passes, will be

$$L_{coil}(w,y,n) = N(w,y,n)\cdot(y+1mm)\cdot n$$

$L_{coil}(w,y,n)$ is set equal to the length of the condenser in order to maximize contact with the air cooled surface. Doing so and solving for w for varying values of y and n and setting a design limit of ΔP =1psi, in a specific embodiment, the final design is found to be

n	y [mm]	w [mm]	$L_{channel}$ [m]	N	d[cm]
5	4	0.5	1	4.61	6.9

for a cycle load of 200W.

Further design parameters can take into account the pressure losses from refrigerant flowing through the helical channels. The pressure loss, ΔP, of the internal flow can be calculated to check that the design does not induce excessive inefficiencies to the thermodynamic cycle of the cooling device. Similarly to the heat transfer coefficient, ΔP can be a function of the flow conditions, the cross sectional geometry, and the length of the tube. Correlations to model the pressure loss may be found in McDonald, A.T., and Fox, R.W., *Introduction to Fluid Mechanics,* John Wiley and Sons, Inc. (2000), which is hereby incorporated herein by reference. Pressure loss can be reduced by reducing the length of the duct, since pressure loss and length can be directly proportional. The length of the duct may be reduced by dividing the flow into multiple ducts. In a specific embodiment, the number of ducts is one continuous channel. In a further embodiment, the number of ducts is 2 or more ducts flowing in parallel.

The fluid that the heat is rejected to can flow through the condenser due to the forces generated by, for example, wind, natural convection, fans, blowers, or compressors. In a specific embodiment, referring to Figure 2, air can be blown into the condenser via, for example, a fan **570**, such that air from air inlet port **3** is blown into the condenser and removes heat from the extended surface features **860**. A fan motor **560** can power the fan **570** having one or more fan blades. One or more of the components of the subject cooling system can be located, at least partially and preferably substantially,

transfer resistance through the separating medium. In a specific embodiment, a parallel channel configuration can be utilized. In a further specific embodiment, the parallel channel configuration can have a separation wall of 1mm and can follow the path of an Archemedian spiral. An archemidian spiral is defined in a parametric coordinate system as:

$$x(t) = A \cdot t \cdot \cos(B \cdot t)$$

$$y(t) = A \cdot t \cdot \sin(B \cdot t)$$

where the constants A and B govern the number of spiral revolutions and the overall diameter of the geometry. One example yields a spiral path as is seen in Figure 3. The path shown in Figure 3 can be used for one fluid, while rotating the path by 180 degrees can provide a path to be used by the second fluid. In other embodiments, other interdigitiated spiral paths can also be utilized.

In a specific embodiment, the path for both fluids can begin on the outer edge of the cylinder and terminate in the center, where both fluids can exit perpendicular to the plane that they are flowing parallel on. Such a design can eliminate abrupt fluid turning points, thus minimizing pressure drop. Thin separation walls can be used to provide a sufficient length of, for example, approximately 25 inches within the limited area of the evaporator having a diameter of 53 mm. The channel depth can be chosen, using two-phase heat transfer correlations as a guide, to maximize the heat transfer area available for both fluids and meet the heat exchange rate requirements of the evaporator. In a further specific embodiment, a channel depth of about 8 mm can be used with an evaporator having 25 inch long fluid path with an evaporator diameter of 53 mm.

A specific embodiment of the subject compact vapor compression cooling system, shown in Figure 4 and 5, can employ a compact assembly which reduces empty space. Open space can be utilized for airflow to remove heat from the cooling system. A cylindrical or spherical shape enhances the surface area of several of the components of the vapor compression cycle so as to reduce the volume of the system. In a specific embodiment, the cylindrical shape can allow for ease of assembling of the components, along with enhanced surface area to volume ratios of the components. Each of the components can be designed into cylindrical shapes, with similar diameters. The

substantially tubular shaped condenser.

Claim 5 (original):

The apparatus for cooling according to claim 4,

wherein the flow of the first external fluid is substantially from the first end of the condenser to the second end of the condenser.

Claim 6 (original):

The apparatus for cooling according to claim 1,

wherein the compressed refrigerant from which heat is removed by the first external fluid in thermal contact with the heat transfer surface flows through the condenser such that the flow of the compressed refrigerant is substantially parallel to the heat transfer surface.

Claim 7 (withdrawn):

The apparatus for cooling according to claim 4,

wherein the condenser has a cross-sectional shape selected from a group consisting of: rectangular, polygonal, square, hexagonal, peanut, and oval.

Claim 8 (original):

The apparatus for cooling according to claim 4,

wherein the condenser has a substantially circular cross-sectional shape.

Claim 9 (currently amended):

~~The~~ An apparatus for cooling ~~according to claim 4~~, comprising:

a condenser having a heat transfer surface, wherein the condenser acts as a heat exchanger so that heat is removed from a compressed refrigerant by a first external fluid in thermal contact with the heat transfer surface of the condenser;

an expansion device, wherein the expansion device receives refrigerant from the condenser, wherein the refrigerant received from the condenser is expanded through the expansion device;

an evaporator, wherein the refrigerant exiting the expansion device flows through the evaporator, wherein the evaporator is in thermal contact with a heat source, wherein the refrigerant absorbs heat from the heat source as the refrigerant passes through the evaporator;

a compressor, wherein the compressor receives the refrigerant exiting from the evaporator, wherein the compressor compresses the refrigerant received from the evaporator, wherein the compressed refrigerant exits the compressor and flows into the condenser; and

a means for flowing the first external fluid across the heat transfer surface of the condenser, wherein the flow of the first external fluid is substantially parallel with the heat transfer surface of the condenser,

wherein the condenser comprises a second surface, wherein the second surface is substantially parallel to the heat transfer surface, wherein the condenser has a substantially tubular shape having a first end and a second end, wherein the heat transfer surface is on the exterior side of the substantially tubular shaped condenser and the second surface is on the interior side of the substantially tubular shaped condenser, and wherein a volume is formed by the second surface of the substantially tubular shaped condenser,

wherein the compressor is positioned substantially within the volume formed by the second surface of the condenser.

Claim 10 (previously presented):

The apparatus for cooling according to claim 9,

wherein the evaporator is positioned substantially within the volume formed by the second surface of the condenser.

Claim 11 (previously presented):

The apparatus for cooling according to claim 10,

wherein the expansion device is positioned substantially within the volume formed by the second surface of the condenser.

wherein each channel of the pair of parallel channels substantially follows the path of a corresponding archemidian spiral.

19. The apparatus for cooling according to claim 1,

wherein the condenser is a gas to vapor heat exchanger, where the vapor is hotter than the gas.

20. The apparatus for cooling according to claim 1,

wherein the condenser is a liquid to vapor heat exchanger, wherein the vapor is hotter than the liquid.

21. The apparatus for cooling according to claim 1,

wherein the expansion device is throttling valve.

22. The apparatus for cooling according to claim 1,

wherein the temperature of the liquid refrigerant liquid is reduced to at least to corresponding saturation temperature upon exiting the expansion device.

23. The apparatus for cooling according to claim 2,

wherein the second external fluid is a liquid.

24. The apparatus for cooling according to claim 2,

wherein the second external fluid is a gas.

25. The apparatus for cooling according to claim 1,

wherein the compressor comprises a positive displacement means such that a first volume of refrigerant vapor enters the positive displacement means and is compressed such that a second volume of compressed refrigerant vapor exits the positive displacement means, wherein the second volume is smaller than the first volume.

26. The apparatus for cooling according to claim 25,

wherein the positive displacement means comprises a mechanism selected from the group consisting of: a piston, a sliding vane, a screw, and a scroll.

27. The apparatus for cooling according to claim 25,

wherein the positive displacement comprises a rotary lobe,

wherein the rotary lobe comprises a substantially triangular shape rotor which spins on an eccentric shaft, wherein the rotor rotates inside an epiterchoid chamber.

28. The apparatus for cooling according to claim 27, further comprising:

one or more spring loaded tip seals on the rotor.

29. The apparatus for cooling according to claim 27, further comprising:

one or more spring loaded face seals on the rotor.

30. The apparatus for cooling according to claim 27, further comprising:

a means for driving the shaft which spins the rotor.

31. The apparatus for cooling according to claim 27, further comprising:

a motor, wherein the motor drives the shaft which spins the rotor.

32. The apparatus for cooling according to claim 31, further comprising:

a motor controller, wherein the motor controller controls the speed of the motor to adjust the rate of compression cycles..

33. The apparatus for cooling according to claim 32,

wherein the motor controller adjusts the rate of compression cycles to match the cooling load.

34. The apparatus for cooling according to claim 1,

a motor, wherein the motor drives the shaft which spins the rotor.

Claim 32 (previously presented):

The apparatus for cooling according to claim 31, further comprising:

a motor controller, wherein the motor controller controls the speed of the motor to adjust the rate of compression cycles.

Claim 33 (original):

The apparatus for cooling according to claim 32,

wherein the motor controller adjusts the rate of compression cycles to match the cooling load.

Claim 34 (original):

The apparatus for cooling according to claim 1,

wherein the first external fluid is air.

Claim 35 (original):

The apparatus for cooling according to claim 1,

wherein the first external fluid is water.

Claim 36 (currently amended):

~~The~~ An apparatus for cooling ~~according to claim 1~~, comprising:

a condenser having a heat transfer surface, wherein the condenser acts as a heat exchanger so that heat is removed from a compressed refrigerant by a first external fluid in thermal contact with the heat transfer surface of the condenser;

an expansion device, wherein the expansion device receives refrigerant from the condenser, wherein the refrigerant received from the condenser is expanded through the expansion device;

an evaporator, wherein the refrigerant exiting the expansion device flows through the evaporator, wherein the evaporator is in thermal contact with a heat source, wherein the refrigerant absorbs heat from the heat source as the refrigerant passes through the evaporator;

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formed between the first element and the second element for the flow of the compressed refrigerant vapor through the condenser, wherein an interior surface of the first element is the second surface of the condenser and an exterior surface of the second element in the heat transfer surface of the condenser.

59. The apparatus for cooling according to claim 58,

wherein the duct is a helical duct.

60. The apparatus for cooling according to claim 58,

wherein a plurality of ducts are formed between the first element and the second element such that the plurality of ducts are parallel with each other.

61. The apparatus for cooling according to claim 6,

wherein the flow of the compressed refrigerant is substantially perpendicular to the flow of the first external fluid

62. A condenser, comprising:

a heat transfer surface, wherein the condenser acts as a heat exchanger so that heat is removed from a compressed refrigerant by a first external fluid in thermal contact with the heat transfer surface of the condenser; and

a means for flowing the first external fluid across the heat transfer surface of the condenser, wherein the flow of the first external fluid is substantially parallel with the heat transfer surface of the condenser.

63. The condenser according to claim 62,

wherein the condenser acts as a heat exchanger so that heat is removed from compressed refrigerant vapor by the first external fluid in thermal contact with the heat transfer surface of the condenser such that the temperature of the compressed refrigerant vapor decreases below the saturation temperature of the refrigerant and the refrigerant vapor condenses to liquid refrigerant, wherein compressed refrigerant vapor flows into

Claim 94 (withdrawn):

The condenser according to claim 93,

wherein the duct is a helical duct.

Claim 95 (withdrawn):

The condenser according to claim 93,

wherein a plurality of ducts are formed between the first element and the second element such that the plurality of ducts are parallel with each other.

Claim 96 (new):

The apparatus according to claim 1,

wherein the condenser comprises a dividing wall having an interior surface and an exterior surface, wherein the interior surface is in thermal contact with the compressed refrigerant and the exterior surface is the heat transfer surface.

Claim 97 (new):

The apparatus according to claim 4,

wherein the second surface of the condenser comprises an enhanced surface geometry, wherein the enhanced surface geometry enhances heat removal by the first external fluid.

Claim 98 (new)

The apparatus according to claim 98,

wherein the first external fluid is ambient air, wherein the enhanced surface geometry of the heat transfer surface of the condenser comprises a plurality of extended surface features, wherein the plurality of extended surface features increase the surface area of the heat transfer surface of the condenser compared with a base surface area of the heat transfer surface of the condenser.

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